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DESIGN STUDIES OF CONDENSERS AND RADIATORS FOR CESIUM AND POTASSIUM VAPOR CYCLE SPACE POWER PLANTS

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FOREWORD

This report summarizes design studies of radiators and condensers made as part of an analytical comparison of cesium and potassium as working fluids for Rankine cycle space power plants. The work was conducted by the Oak Ridge National Laboratory for NASA under AEC Interagency Agreement 40-98-66, NASA Order W-21,353 under the technical management of A. P. Fraas of the Oak Ridge National Laboratory. Project management for NASA was performed by S. V. Manson of NASA Headquarters.

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A. P. Fraas

Abstract

The information available on the design and development of condensers and radiators for space power plants has been applied to the preparation of reference designs for condensers and radiators for two 300 kw(e) nuclear electric space power plants employing cesium and potassium, respectively. There is no significant difference in the size or weight of the radiators or the condensers for the two working fluids.

INTRODUCTION

The Oak Ridge National Laboratory has undertaken an analytical comparison of cesium and potassium as working fluids for Rankine Cycle Space Power Plants for the National Aeronautics and Space Administration (AEC Interagency Agreement 40-98-66, NASA Order W-12,353). This report is one of a series that have been prepared as a part of that study and presents the background considerations and design calculations for the condensers and radiators for the reference design power plants.[1]

The first step in the study was a review of the majority of the fairly complete space radiator design studies that had been prepared previously[2-10] together with pertinent information on heat transfer and fluid flow,[11-15] meteoroid incidence and penetration,[16-27] and the properites, compatibility, and fabricability of materials.[28-33] With this information at hand, heat transfer matrix geometries and overall configurations were compared, particular geometries were chosen, and detail designs were evolved.

DESIGN CONDITIONS

The basic work statement from NASA specified that the study should be based on a three-loop system with a turbine inlet temperature of 2150°F (with 25°F of superheat) and a condenser temperature of 1330°F. The thermodynamic and aerodynamic analyses covered in connection with the turbine studies[34] indicate that the difference between the cesium and the potassium cycle efficiencies is so small that, to a first approximation, there is no difference in the design requirements for the radiators for the two systems. There are, of course, substantial differences in the vapor volume flow rates into the cesium and potassium condensers so that these require separate treatments.

In specifying the heat load on the radiator, the net electrical power output of 300 kw specified by NASA together with the overall Rankine cycle system efficiency of 17% indicated by the turbine design studies[34] define the amount of waste heat from the Rankine cycle to be dissipated by the radiator. An additional allowance must be made for the electrical power input to the pumps in the primary and radiator circuits. Studies by Pratt & Witney, AiResearch, General Electric, and ORNL have indicated that a well proportioned system results if the electrical power input to the pumps is about 10% of the net electrical output and that about two-thirds of this should be employed in the radiator circuits. Design studies of the pumps that might be used (presented in a companion report[35]) indicate that the pump efficiency will run around 20%. Thus the electrical power available for the pumps in the radiator circuits will be about 20 kw, and the useful pumping work will be about 4 kw.

A number of different reliability studies[3,8] have indicated that from four to eight separate radiator circuits will give close to the maximum reliability obtainable when allowances are made for outages caused by equipment failures as well as meteoroid punctures. Four circuits give a high probability that the system output would not be reduced by more than 25% as a result of a malfunction in a component in a radiator circuit or a meteoroid puncture of a radiator, and this number of circuits was chosen for the subject study.

RADIATOR TUBE, FIN, AND ARMOR CONFIGURATIONS

There now seems to be widespread agreement that the radiator of a nuclear space power plant should be designed to conform to the envelope of the launch vehicle and contribute to it structurally.[3,5,7] This implies that the radiator should be a conical or cylindrical shell with the tubes running longitudinally in planes through the axis. To simplify the comparisons, a 10-ft-diam cylindrical radiator suitable for installation on a Titan-III launch vehicle has been assumed for the purposes of this study.

Form of Meteoroid Armor

The three principal ways of arranging the tubes, metal fins, and armor that have been seriously considered in recent years are shown in Fig. 1 (Refs. 3,5,7). No meteoroid armor is required on the back sides of the tubes because the portion of the radiator on the opposite side coupled with closures at the ends will act as a bumper and protect it.[24] A hypervelocity projectile is fragmented or vaporized in passing through even a thin sheet and would be dispersed into a cloud of vapor or particles by the time it had passed a foot or more beyond a fin or reflector that would act in this fashion. While configuration A at the top has been used in many design studies, this meteoroid armor geometry provides only partial protection because the meteoroid flux is roughly isotropic, and meteoroids are as likely to strike at an acute angle to the radiator surface as they are to come in normal to the radiator envelope. In the event of an oblique impact, the fin is too close to the tube to be effective as a bumper. Further, the crater depth resulting from an impact is essentially independent of the angle of incidence down to angles of about 12 deg. In view of these factors, the armor geometry should be similar to that shown in configurations B or C.

Heat Pipes as Fins

Heat pipes are such potent conductors of heat that a number of proposals have been made to use them as fins for space radiators. One such geometry is shown in Fig. 2.[5] The proponents of this configuration point out that if extensive compartmentalization were employed a meteoroid puncture of one compartment would not interfere with the heat transfer

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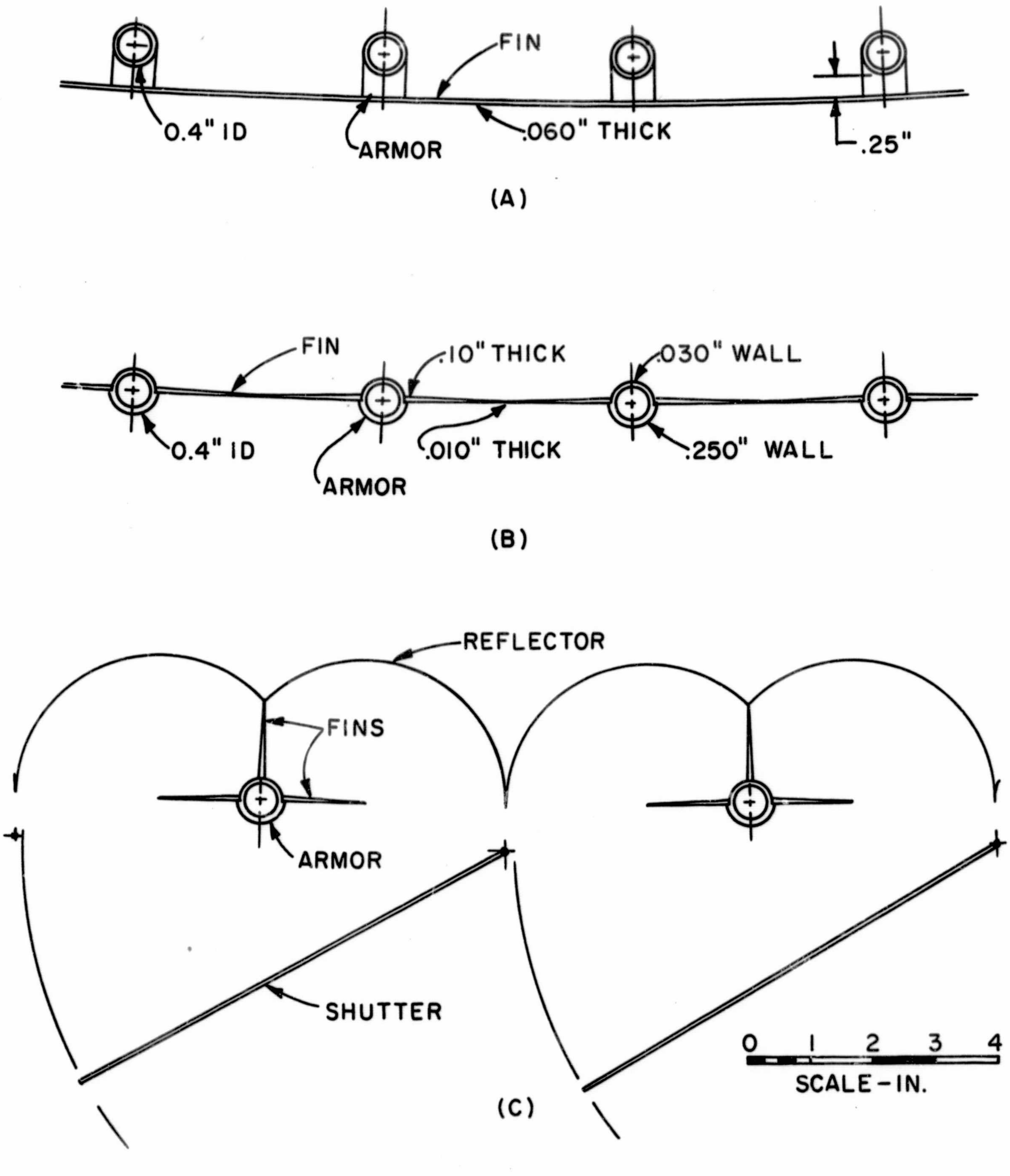

Fig. 1. Three Typical Tube-Fin-Armor Configurations for Radiators for Space Vehicles.

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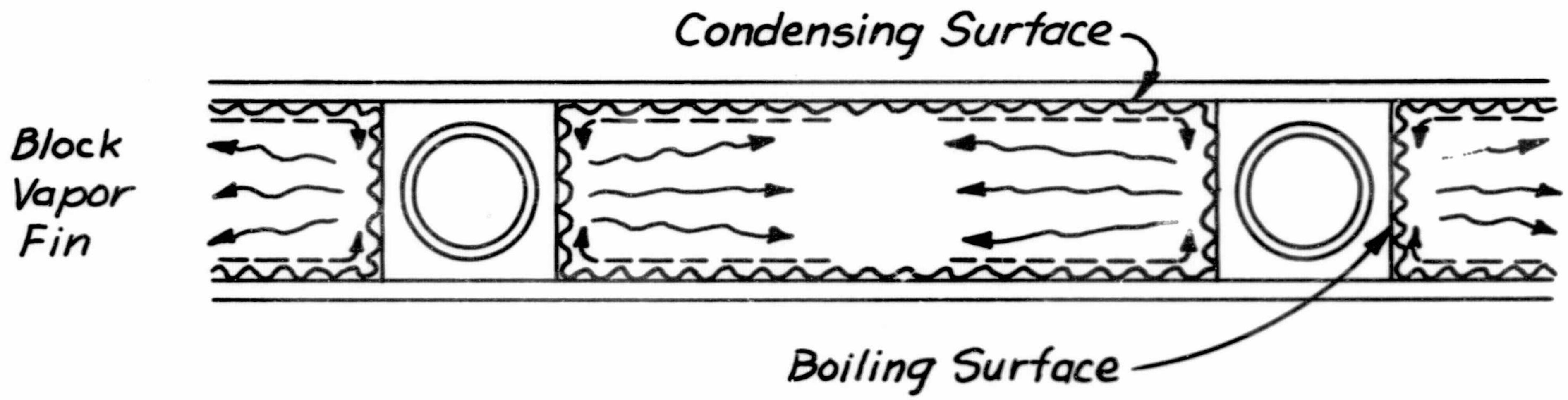

Fig. 2. Heat Pipes Used as Fins for Space Radiators.

performance of the others. A thin-walled sheet would be used which would give a lighter construction than would result for thick, solid fins. However, there are some serious problems associated with the construction of Fig. 2. For one thing, the system must operate during a check-out on the ground with atmospheric pressure surrounding the radiator and a vacuum of a few millimeters of mercury within the heat pipe region. This would require both much heavier sheet on the surfaces than would otherwise be necessary and would also require the introduction of numerous bulkheads. Note, too, that compartmentalization is required to allow for the large axial temperature drop along the length of the tube. An even more serious problem stems from the need for a high degree of leak-tightness. The assembly would probably have to be fabricated by brazing, and this in turn will require that over 1000 ft of brazed seam be leak-tight even if there were no compartmentalization, and several thousand feet of seam if the unit were compartmentalized. Compartmentalization would also greatly increase the difficulties and cost associated with evacuating the heat pipe regions and charging them with potassium. The leak-tightness problem is rendered even more difficult by thermal stresses. Severe shear stresses would be induced as a consequence of the non-uniform temperature distribution that would be associated with any irregularities in flow distrution or which would result from leaks from any compartments. These stresses would be particularly serious at the corners where the seams would stiffen the matrix and yet brazing would be most likely to cause local embrittlement.

If any compartments were to leak, the local temperature of the sheet metal in the space between adjacent tubes would drop sharply and generate severe thermal stresses. These would be likely to be sufficiently severe along the bulkheads between compartments to cause leaks to develop as a consequence of low cycle fatigue effects under changing load conditions, a gradual increase in the number of compartments leaking would occur, and deterioration of the radiator would result. Worse, cracks in the skin would be likely to propagate and form cracks in the tubes, and these would lead to leakage of NaK from the system. Another problem is that low vapor pressures lead to poor performance of the heat pipe at temperatures below 700°F. As a consequence, under part-load conditions the temperature distribution in the outer skin would be poor with the temperature of the skin midway between tubes running much below that of the tubes so that severe thermal stresses, warping, and possibly cracking could result.

The above considerations make it difficult indeed to devise a heat pipe configuration that will meet all of the requirements including the need for very high reliability. In view of the absence of any appreciable amount of radiator test experience to support the heat pipe proposal, it seemed best to employ solid fins rather than a heat pipe system such as that of Fig. 2.

Fabrication and Thermal Stress Problems

Configurations A and B have the advantage that they are conventional and, on the surface, are easy to analyze structurally. Actually, under operating conditions, deviations from ideality in the coolant flow and temperature distribution can lead to serious thermal stresses and warping. Further, large panels of this type are difficult to fabricate since they are difficult to jig for the brazing operation and small amounts of distortion during brazing will lead to a poor bond between the fins and the tubes.

Configuration C with the reflector is unconventional, but avoids the above problems. Several units of this type ranging from 30 kw to 360 kw in heat rejection capacity have been operated in liquid metal systems for a total of over 12,000 hr.[2,9] A unit of this type (10 ft in diam and 14 ft in height) has been built, and is shown in Fig. 3. The brazed joints are excellent, and one need only strike it a sharp blow with his fist to convince himself that the structure is rugged, strong, and stiff. Not only can good brazed joints be obtained,[33] but tapered fins can be used readily, thus reducing their weight. The tubes are sufficiently flexible to accommodate differential thermal expansion by column buckling, yet stiff enough so that, when bonded by polyurethane foam to the reflector and a windshield formed by closing the shutters shown, they would form a strong, vibration-resistant structure for launching. The polyurethane foam would vaporize when heated in the vacuum of space during start-up.

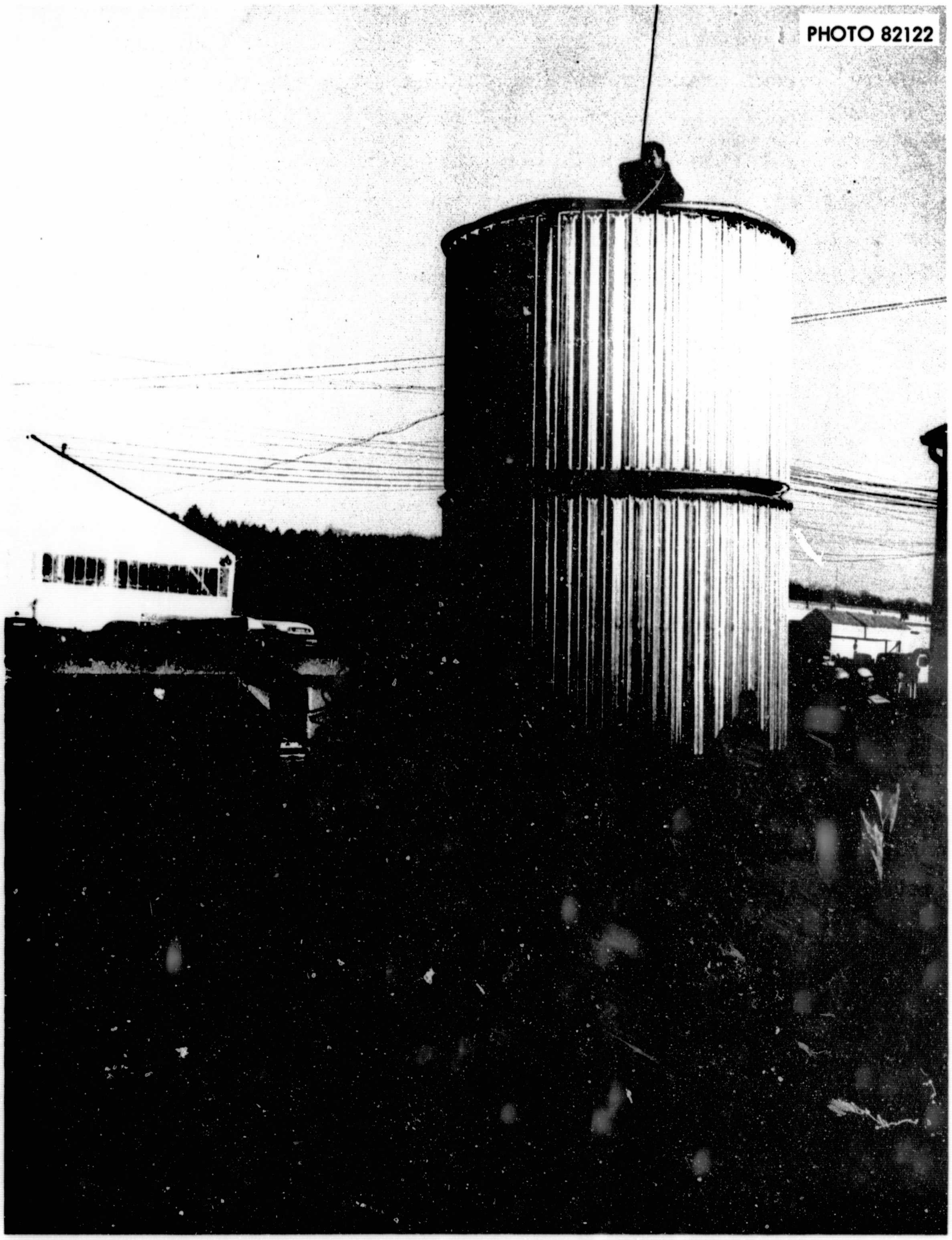

Fig. 3. Full Scale Radiator Built with the Configuration of Fig. 1-C with Finned Tubes and Reflectors.

Effects of Tube Spacing on Radiator Weight

The most important characteristic of the reflector and finned tube configuration of Fig. 1-C is that it gives a much lower weight than the other two by making the rear surfaces of the fins nearly as effective as the front. Since the fin weight varies as the square of the fin height for a given heat flux and fin efficiency, the fin (and armor) weight for a given tube diameter and tube spacing are much lower for the reflector arrangement than for the other two. Of course, the minimum value for the sum of fin and armor weights is obtained with a closer tube spacing for the upper two configurations of Fig. 1 than for the lowest that employs a reflector. The effects of tube spacing on the weight of configurations B and C of Fig. 1 are shown in Fig. 4. Note that the minimum radiator surface weight obtainable with configuration C is about half that for configuration B. The calculations for Fig. 4 were made for the same heat flux from the radiator envelope, the same tube diameter, the same fluid temperature in the tubes, and the same probability for meteoroid penetration in both cases. The detailed calculations are summarized in Table 1. The design conditions were almost the same as those specified in Table 2 for the reference designs of this study.

Choice of Heat Transfer Matrix Geometry

A number of vital points stand out in a review of the above discussion, that is,

1. The finned tube and reflector gives the lightest matrix of the three principal finned tube geometries that appear attractive, and is free of severe thermal stress problems.

2. While the use of heat pipes as fins appears intriguing, the structures proposed are inherently subject to severe thermal and pressure stresses and hence do not appear to be reliable.

3. The severe difficulties inherent in brazing beryllium, its brittleness, and its embrittling effects on stainless steel would lead to high costs and reduced structural integrity. Copper fins and stainless steel armor have been fabricated readily to give a sound ductile structure

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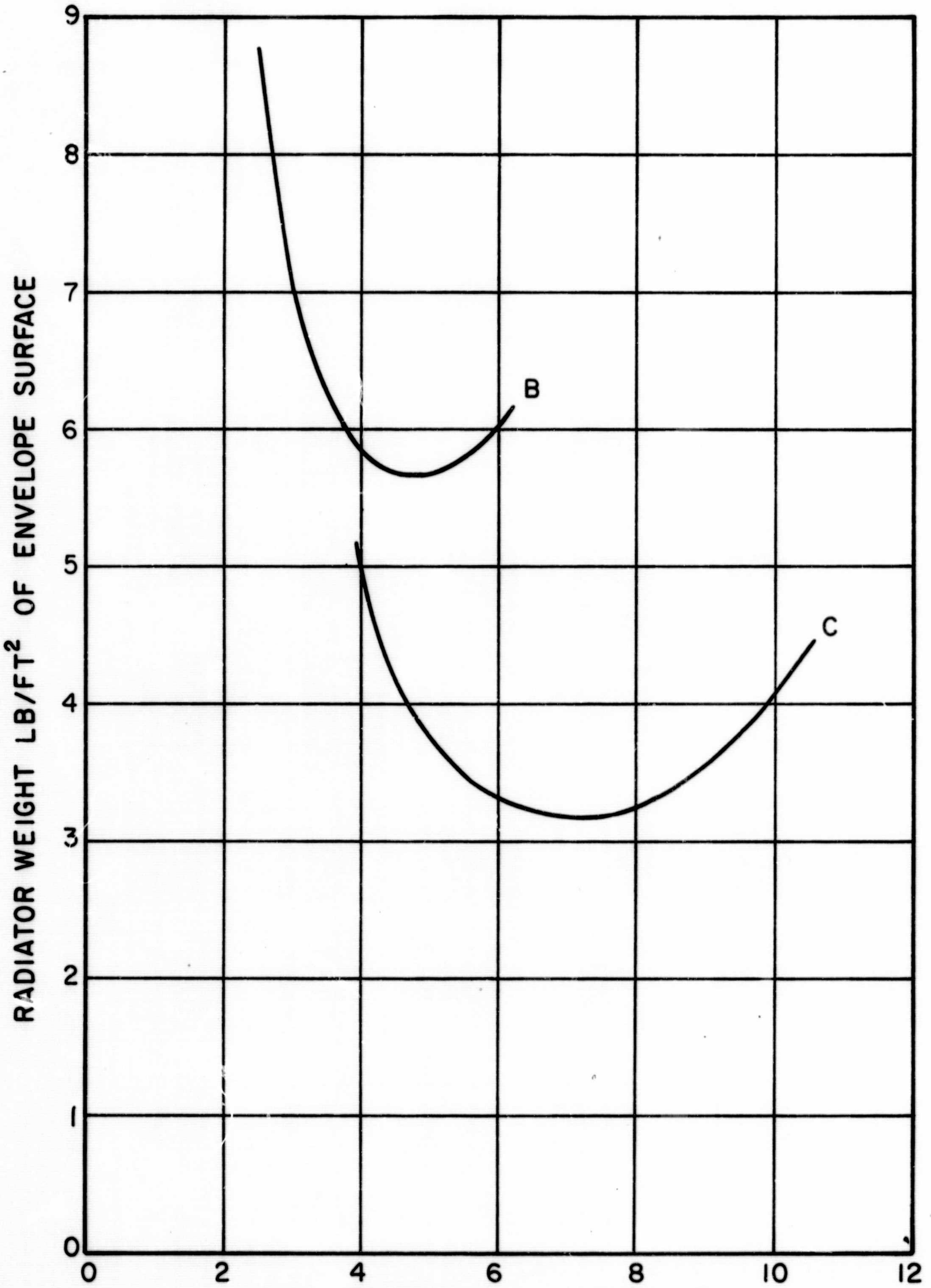

Fig. 4. Effects of Tube Centerline Spacing on the Radiator Weight Per Square Foot of Surface Area of the Radiator Envelope.

Table 1. Effects of Tube Spacing on the Radiator Weight for Configurations B and C of Fig. 1

T = 1380°F; ϵ = 0.92; Fin + tube efficiency = 0.773; Tube ID = 0.38 in.; Fin T = b $(x/w)^2$ copper fins

Configuration B				
Tube centerline spacing, in.	2.5	3.0	4.0	6.0
Relative vulnerable area	1.67	1.36	1.0	0.65
Armor thickness, in.	0.246	0.234	0.217	0.194
Armor weight, lb/ft	1.63	1.50	1.37	1.19
Armor weight, lb/ft^2 (total surface)	7.82	6.0	4.15	2.38
Fin height, in.	1.03	1.28	1.78	2.78
Fin efficiency, %	72.5	73.4	74.3	75.2
Fin root thickness, in.	0.027	0.044	0.090	0.228
Fin weight, lb/ft^2 (total surface)	0.572	0.83	1.48	3.5
Weight of rear portion of tubes, lb/ft^2	0.36	0.28	0.21	0.14
Total weight, lb/ft^2	8.75	7.1	5.84	6.02
Configuration C				
Tube centerline spacing, in.	4.0	6.0	8.0	10.0
Relative vulnerable area	1.0	0.65	0.46	0.35
Armor thickness, in.	0.217	0.194	0.176	0.166
Armor weight, lb/ft	1.37	1.19	1.03	0.96
Armor weight, lb/ft^2 (total surface)	4.15	2.38	1.54	1.20
Fin height, in.	0.615	1.03	1.45	1.86
Fin efficiency, %	76.8	79.3	80.0	81.2
Fin root thickness, in.	0.025	0.082	0.173	0.306
Fin weight, lb/ft^2 (total surface)	0.67	0.80	1.60	2.79
Weight of rear portion of tubes, lb/ft^2	0.21	0.14	0.10	0.08
Total weight, lb/ft^2	5.03	3.32	3.24	4.07

Table 2. Summary of Data and Design Calculations for the Reference Design Radiator

	Item	Source	Value
1)	Cycle efficiency, %		17
2)	Vapor temperature into condenser, °F		1330
3)	Condensate temperature out of condenser, °F		1300
4)	NaK temperature into condenser, °F		1200
5)	NaK temperature out of condenser, °F		1300
6)	Heat load, kw	300/1	= 1430
7)	Heat load, Btu/hr	3413 × 6	= 4,890,000
8)	Electrical power input to NaK pumps, kw		20
9)	Efficiency of NaK pump, %		20
10)	NaK flow rate (total), lb/hr	7/.25(3 − 4)	= 195,300
11)	NaK flow rate (total), ft^3/hr	10 / 13	= 4,140
12)	NaK flow rate (total), ft^3/sec	11 /360	= 1.15
13)	NaK density, lb/ft^3		47.2
14)	NaK pressure drop (total), psi	8 9 × 778 × 3413/ 11 × 14,400	= 17.8
15)	Mean radiator temperature, °F	(3 + 4)/2	1250
16)	Nominal radiation heat flux, Btu/hr		14,300
17)	Fin efficiency		.80
18)	Surface emissivity		.92
19)	Reflector efficiency, %		87
20)	Fraction of total surface for reflector		.58

Table 2. (Continued)

	Item	Source	Value
21)	Mean effective heat flux, Btu/hr	(16) (17) (18) [1 − (1 − (19)) (20)]	= 9,680
22)	Surface area required, ft^2	(7) / (21)	= 505
23)	Radiator diameter, ft		10
24)	Tube spacing, in.	(23) /50	= 7.85
25)	Number of tubes per bank		48
26)	Radiator height, ft	(22) / (23) π	= 16.0
27)	Number of tube banks		2
28)	Tube length, ft		8.0
29)	Pressure drop in tubes, psi	(14) /4	= 4.4
30)	Dynamic head, psi		1
31)	NaK velocity for item 30, ft/sec		14.1
32)	Tube flow passage area required, $in.^2$	(12) × 144/96 × 31	= .127
33)	Tube flow passage area chosen, $in.^2$		.151
34)	Tube I.D., in.		.44
35)	Tube O.D., in.		0.50
36)	Fin span, in.		3.2
37)	Fin height, in.		1.35
38)	Fin base thickness for fin efficiency = 80, in.		.16
39)	Vulnerable surface area, ft^2	(26) (25) (35) π/24	= 51
40)	Meteoroid armor thickness required, in.		0.092

Table 2. (Continued)

	Item	Source	Value
41)	Inlet manifold I.D., in.		1.88
42)	Inlet manifold O.D., in.		2.00
43)	Outlet manifold I.D., in.		1.38
44)	Outlet manifold O.D., in.		1.50
45)	Tube weight, lb		115
46)	Armor weight, lb		138
47)	Fin weight, lb		700
48)	Reflector weight, lb		180
49)	Manifold weight, lb		100
50)	Shutter (shroud) weight, lb		120
51)	Total radiator dry weight, lb		1353
52)	Total radiator weight (with NaK), lb		1440

that is almost as light as can be obtained with beryllium fins and armor and is relatively inexpensive.

4. The only space radiator heat transfer matrix geometry that has been tested extensively at temperatures above 1000°F is the finned tube and reflector of Fig. 1-C. (About 12,00 hr of operation have been obtained at ORNL with this type of tube matrix.)

In view of these points, the finned tube and reflector geometry of Fig. 1-C was chosen for the subject study. This choice has further advantages not evident in the above discussion in that it gives a heat transfer matrix that has relatively little vulnerable area, requires relatively little armor, and hence its weight is insensitive to the choice of armor material.

RADIATOR REFERENCE DESIGNS

Choice of Tube Diameter

Once a basic heat transfer matrix geometry and radiator configuration are chosen, the size and weight of the radiator can be estimated in a fairly straightforward fashion. The major remaining question relative to the geometry is that of tube diameter and length. If only a small element of the radiator is considered, studies show that the specific weight drops somewhat with a reduction in the tube diameter. For the radiator size required here, 1/2 in. OD straight tubes were found to give a well proportioned unit with a single central ring header feeding a bank of tubes on either side with a ring-shaped outlet header at either end as in Fig. 5. While some studies of other designers have favored somewhat smaller tube diameters, the increased NaK pressure drop per unit of length would have required doubling the number of headers (and tube-to-header joints) to avoid excessive pumping power. Further, the vulnerable surface area depends on the tube OD, and since the meteoroid armor thickness required is about 0.15 in., there is little to be gained by reducing the tube ID below about 0.4 in. This is especially true if allowances are made for the ring headers because their vulnerable area increases as more are required for use with smaller diameter, shorter

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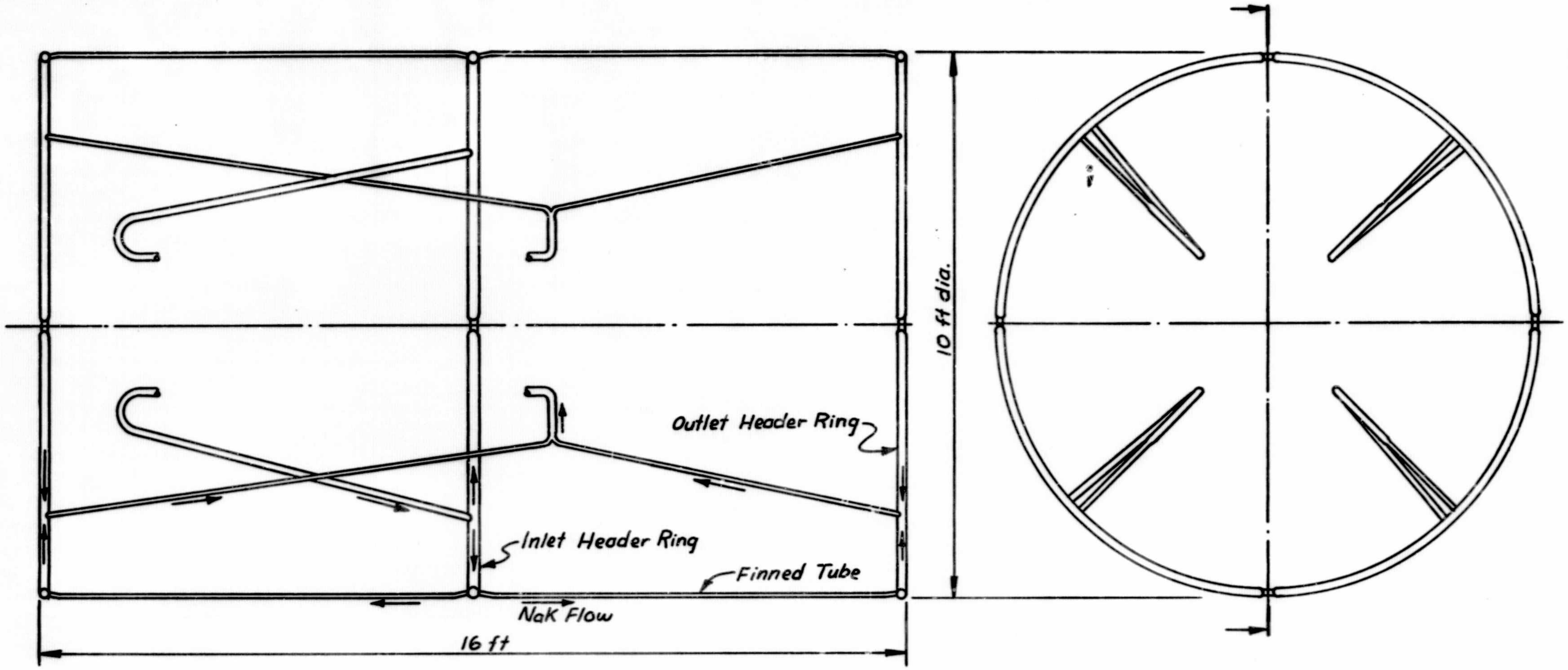

Fig. 5. Configuration Chosen for Reference Design Radiators.

tubes. The proportions shown in Fig. 5 yield a sound overall structure with tube and manifold length-diameter ratios small enough for good structural strength, and a tube length-diameter ratio large enough to accommodate differential thermal expansion between tubes by column buckling.

Tube Wall Thickness

The basic tube wall thickness was chosen to be 0.030 in. on the basis of extensive endurance test experience at ORNL with high temperature heat exchangers.[36,38] This experience has shown that individual grains sometimes grow to a size of about 0.010 in., and, since there may be imperfections in individual grain boundaries, it is important to make the tube wall thickness at least twice that of the largest grains, that is, at least 0.020 in. thick.[36] The thickness was increased to 0.30 in. to increase the system integrity because the resulting weight penalty was only about 20 lb. (There has not been a single failure in tube walls running 0.020 in. to 0.030 in. in over 70,000 hr of ORNL heat exchanger tests at temperatures over 1000°F.)

Choice of Materials

Copper and beryllium have been the principal materials considered for use as fins in space radiators designed for operation in the 1000°F to 1500°F temperature range. From the fin weight standpoint, the logical figure of merit is the ratio of the thermal conductivity to the density. This parameter at 1000°F is 0.51 for beryllium and 0.40 for copper.

Fabrication considerations are vital in a realistic design. Beryllium is brittle at room temperature, is very difficult to braze, and tends to embrittle the stainless steels and the refractory alloys through the formation of beryllides. (Its effects are similar to those of boron, to which it is chemically similar.) These problems are accentuated by the fact that large thermal stresses are generated because the coefficient of thermal expansion of beryllium is only 2/3 that of stainless steel and is double that of the refractory alloys. Further, beryllium base stock is expensive, costing about $50/lb for slabs or heavy plate. Fabrication costs are high because of its brittleness and toxicity. Thus simple fabricated parts of beryllium cost about $100/lb. This compares with about $1/lb for copper which is strong, ductile, has about the same coefficient of thermal expansion as stainless steel, and is easily brazed to stainless steel to give a strong, ductile joint.

While oxidation will not be a problem in space, extensive development testing will be required, and hence it is advantageous to choose materials that can be operated in air at 1000°F to 1500°F. Although copper oxidizes at this temperature, excellent protection can be provided by applying a thin layer of stainless steel cladding. Some 45,000 hr of operation at over 1000°F have been obtained at ORNL with radiators having copper fins clad with stainless steel.[2,37] Development tests could be conducted with the stainless steel cladding, and then this could be omitted from units to be launched in order to save weight.

Low-density materials such as aluminum, beryllium, and graphite are somewhat more effective for meteoroid armor on a weight basis than structural materials such as steel or refractory metal.[5] Aluminum has too low a melting point for the application at hand. In turning to the others, ideally the weight advantage of the low-density materials is less than a factor of two, and practically this is largely offset by the problems cited above, that is, brittleness, differential thermal expansion, thermal stresses, and fabrication problems.

Meteoroid Armor Thickness

Several different approaches to the estimation of the thickness of the armor required to provide protection from meteoroids have been employed. The method used was based on information presented in Refs. 22 and 23 as outlined in Ref. 39. This method was chosen for the purposes of the subject study, and the chart of Fig. 6 was prepared. The values given by this chart are within 14% of those given by relations developed by the NASA Lewis Laboratory.[40] The values are also within 20% of the values used by CANEL in their SNAP-50 work (see CNLM-6293, p. 6-3-1) and for the SNAP-8 system (see NASA Specification No. 417-5). The meteoroid armor was assumed to be stainless steel integral with the tube wall around 180 deg

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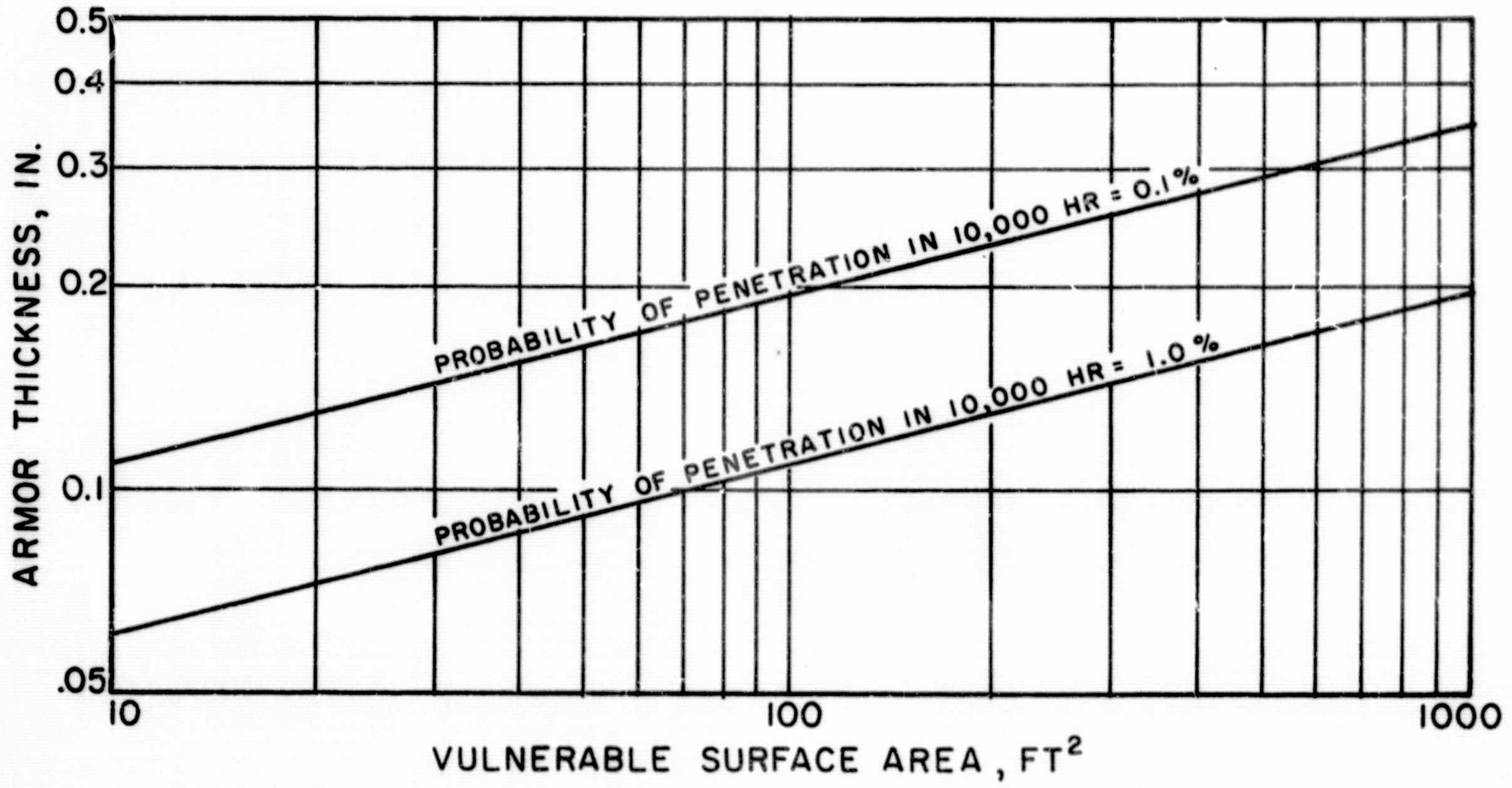

Fig. 6. Effects of Vulnerable Surface Area on the Meteroid Armor Thickness Required for Stainless Steel Radiators.

of the tube perimeter as in Fig. 1-C, and the thickness was chosen to give 0.1% probability of a puncture in 10,000 hr. Figure 7 gives the weight of thick-walled tubing as a function of the ID and the wall thickness. The weight of the armor was taken simply as half of the value given in Fig. 7 for a tube of a given ID and a thickness equal to that of the armor required.

Design Calculations

The design calculations are summarized in Table 2. Note that, where it is not necessarily obvious, the source of the numerical value for each item is indicated in the central column. To save space and facilitate checking of the calculations, circled numbers have been used as symbols to indicate the number of the line from which a numerical value was obtained for use in the equation. However, most of the steps in the table are obvious or stem directly from considerations discussed at some length above.

In reviewing the weights of the various components in the lower portion of the table it is interesting to note that by far the largest item is the weight of the fins which represent a little over half the total weight of the radiator. If beryllium had been employed in place of copper, the fin weight might have been reduced 20%, which would have yielded a reduction in overall radiator weight by about 10%. It is doubtful that the reduction in the system integrity implicit in such a change would be justified by the savings in weight of around 140 lb. It is also interesting to note that, if the manifold and shutter weights are deducted, the total radiator weight would be 1130 lb, or a little less than 4 lb/kw of net electrical output. Specific weights as low as 2.5 lb/kw have been estimated by other organizations for essentially similar operating conditions, but these have been based on more optimistic assumptions such as higher allowable probabilities of meteoroid penetration and the use of beryllium for both fins and armor. Information readily available on these designs from other organizations is not sufficiently detailed to permit a thorough check, but it appears that if allowances were made for all these differences the results would be consistent, and

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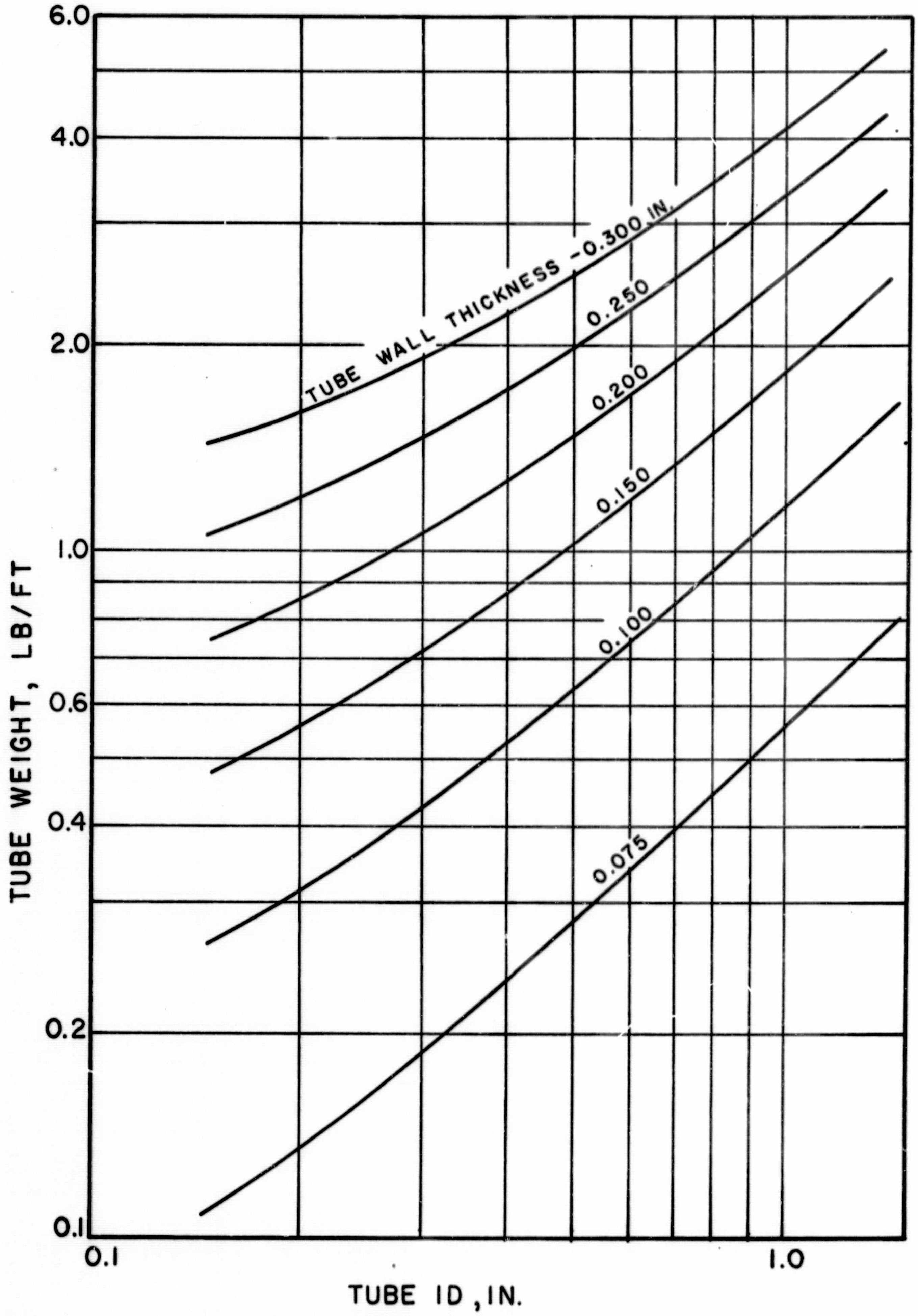

Fig. 7. Weight of Thick-Walled Stainless Steel Tubing.

the reflector and finned tube arrangement of Fig. 1-C would give the lowest specific weight.

GENERAL DESIGN CONSIDERATIONS FOR CONDENSERS

Many different requirements must be accommodated in the design of a condenser if a well-proportioned unit is to be obtained and the unit is to lend itself well to intergration in the complete power plant system. In addition to the obvious heat transfer and fluid flow considerations, allowances must be made for such subtle problems as those associated with zero-g operation, part-load performance, fabrication, and thermal stresses.

Zero-g Operation

The control of free liquid surfaces under zero-g conditions presents a complex set of problems for which there is very little background of engineering experience. Unless a space vehicle is spun to induce an artificial gravitational field, the principal forces acting on droplets of condensate will be surface tension and fluid dynamic forces. Thus the condenser must be designed so that these forces will act to move the condensate along the tube wall to the outlet in a continuous, dependable fashion, or otherwise the vapor space in the condenser will load up with liquid and the system will not function properly.

One way of assuring condensate flow through a condenser under zero-g conditions is to employ a jet condenser in which a subcooled jet of liquid is injected at a high velocity coaxially with the vapor stream into a converging channel. The momentum of the liquid and vapor suffice to carry the stream through the converging region where condensation takes place, and a bubble-free liquid stream emerges sufficiently subcooled to assure freedom from vapor bubbles. Unfortunately, this approach requires that the liquid jet operate at an average temperature much below the saturation temperature at the inlet to the condenser, and this entails a substantial weight penalty. Further, a jet condenser system is not isolated from the power conversion system, and a radiator leak would shut down the main turbine.

If a surface condenser rather than a jet condenser is employed in order to maximize the radiator temperature, it is possible to use uniformly tapered condenser tubes so that the velocity will be high throughout the tube length to provide sufficiently high fluid-friction forces to drive the condensate toward the outlet.[11] This approach has been employed very successfully at ORNL, and extensive experience has been obtained with four steam condensers and three potassium condensers designed and built to operate in this fashion.[9] Over 13,000 hr of operating experience have been obtained with the potassium condensers, and excellent scavenging characteristics have been demonstrated in units operating with the tubes mounted horizontally to minimize the effects of gravitational forces. Since the test results have confirmed the analytical design which was based on a procedure developed prior to any test work, it is believed that the analytical procedure is sound and is well suited to the case at hand.

Vapor Volume Flow Rate

During part-load and startup or shutdown conditions the pressure of vapor entering the turbine will be much lower than at the design point. The relationships are complex and depend in part on the control scheme employed, but, to the writer's knowledge, in any of the control schemes that have been proposed there is a tendency for the vapor volume flow rate into the condenser to increase as the load is reduced from full-power conditions. This stems from the fact that the densities of the saturated vapors of potassium and cesium fall off at an increasingly rapid rate as the vapor temperature is reduced. As a consequence, at low loads choking will occur at some point between the turbine outlet and the condenser outlet. The general problem is treated in a companion report on overall system integration,[1] hence it will simply be stated here that there are excellent reasons for designing the vapor passages so that choking will take place at the inlet to the condenser tubes and that these tubes should be designed for an inlet Mach number not greater than about 0.30 at design conditions. This Mach number limitation will make possible good system performance at part load.[1,9]

Choice of Tube Diameter

To a first approximation, as discussed earlier in connection with the same problem in the radiator, the smaller the tube diameter the smaller and lighter the heat transfer matrix, but the greater the number of tube-to-header joints. An additional problem presents itself in the condenser where relatively large vapor volume flows must be handled. Further, if tapered tubes are employed it is desirable to make the minimum tube diameter at the outlet end at least about 0.20 in. to avoid possible difficulties with plugging or undesirable capillary effects. Previous ORNL studies indicate that, to maintain a reasonably uniform vapor velocity along the greater part of the length of the tube, it is desirable to employ a diameter taper ratio of three.[11] From this it follows that a tube inlet ID of 0.60 in. and an outlet ID of 0.20 in. will give a well-proportioned tube. (Extensive experience has been obtained with potassium condenser tubes having about these proportions.[9])

CONDENSER REFERENCE DESIGNS

Following the design precepts outlined above together with the pertinent considerations and design data discussed earlier in connection with radiators, a series of design calculations was carried out and is presented in Table 3. The log mean temperature difference was obtained from the temperature data in Table 2. The condensing heat transfer coefficient was taken as 10,000 Btu/hr·ft^2·°F on the basis of experimental data obtained both by ORNL[9] and by General Electric in tests for NASA.[13] It should be pointed out that the condensing coefficient for liquid-metal vapors is so high that it is difficult to determine experimentally, but the uncertainty is not important since the barrier to heat transfer represented by the tube wall is about as great as that represented by the condensing film coefficient. The condenser tubes were assumed to be refractory metal. (Limited corrosion and mass transfer data obtained at ORNL indicate that it is possible to use stainless steel in the radiator, or heat rejection zone in a system in which the high temperature, or heat addition, portion is built of a refractory metal.[32]) The heat transfer coefficient on the

Table 3. Design Calculations for Potassium and Cesium Shell-and-Tube Condensers

Item	Source	Value	
1) LMTD, °F	Table 2	80	
2) Heat transfer coefficient for vapor, Btu/hr·ft^2·°F		10,000	
3) Heat transfer coefficient for NaK, Btu/hr·ft^2·°F	Ref. 41	15,000	
4) Conductance of tube wall, Btu/hr·ft^2·°F		10,000	
5) Σ 1/u		0.00037	
6) Overall heat transfer coefficient, Btu/hr·ft^2·°F		2700	
7) Q/A, Btu/hr·ft^2		216,000	
8) Heat transfer surface area required (total), ft^2	4,890,000/7	22.6 ft^2	
		K	Cs
9) Vapor specific volume, ft^3	Ref. 34	43.74	5.5691
10) Vapor flow rate, lb/hr	Ref. 34	1.6	6.6
11) Vapor flow rate, ft^3/sec	Ref. 34	66.42	33.628
12) Sonic velocity, ft/sec	Ref. 42 and 43	1435	681
13) Inlet Mach Number		.30	.30
14) Inlet velocity, ft/sec	12 13	430	204
15) Inlet flow area (total), ft^2	11 / 14	0.1543	0.165
16) Tube inlet ID, in.		0.60	0.60
17) Tube inlet ID, in.		.283	.283
18) No. tubes (total)	144 15 / 17	78.6	84
19) No. tubes per condenser unit	Ref. 41, p. 343	19	22

Table 3. (Continued)

Item	Source	Value	
20) Centerline spacing, inlet header, in.		0.76	0.76
21) Inlet header sheet diam, in.	Ref. 41, p. 343	3.80	4.42
22) Tube outlet ID, in.		0.20	0.20
23) Tube outlet OD, in.		0.26	0.26
24) Tube outlet centerline spacing, in.		0.36	0.36
25) Outlet header sheet/diam, in.	Ref. 41, p. 343	1.80	1.92
26) Vapor inlet pipe diam, in.	$\sqrt{36 \; 15 \; 1.5/0.786}$	3.2	3.4
27) Tube length, in.	$144 \; 8 \; 1.05/\pi 0.43 \; 19 \; 4$	33.2	28.7
28) Tube weight, lb	Fig. 7	7.4	7.4
29) Header weight, lb		3.1	4.1
30) Casing weight, lb		6.8	7.8
31) Total dry weight, lb (per unit, 4 required)		17.3	19.3
32) NaK weight		2.0	2.6

NaK side depends more on the tube spacing than on the NaK velocity for the range of interest. ORNL experience in fabricating tube-to-header sheets indicates that the spacing between tubes should be at least 0.10 in. to assure a high integrity, welded, header sheet.[36] Thus the overall heat transfer coefficient was defined by the tube spacing, and this, coupled with the LMTD, defined the surface area required.

The number of tubes required depends on the inlet volume flow rate which was obtained from a companion report on turbine design.[34] This, coupled with the permissable inlet Mach number of 0.30 cited in the previous section, defines the inlet velocity and inlet flow passage area. With the tube inlet ID specified as 0.60 in., it is easy to determine the number of tubes, the header sheet diameter, and the overall tube length. Note that an extra 5% was added to the latter to sub-cool the liquid and thus provide cavitation suppression head for the jet pump used to scavenge the condenser. Note, too, that four condensers are used in parallel and that the number of tubes in each unit must be chosen to suit one of the discrete number of hole patterns that fit well inside a circle.[41]

In estimating the condenser weight the minimum condenser shell wall thickness was taken as 0.10 in. to provide ample resistance to buckling under external air pressure when the system might be evacuated prior to startup or under the low internal pressure conditions that would prevail under startup and low power operating conditions.

A review of the results of the design calculations indicates that there is surprisingly little difference between the cesium and potassium condensers. A few more tubes are required for the cesium condenser but they are somewhat shorter than those required for the potassium condenser. In reviewing the design procedure it is evident that this should, in fact, be the case because the heat transfer coefficients and tube wall conductance are the same in both cases, and these determine the amount of surface area required. Thus, for the same basic tapered tube heat transfer matrix geometry, the only differences to be expected should be in the length and number of tubes rather than in the volume of the heat transfer matrix.

The layout for the potassium condenser is presented in Fig. 8. The differences between the potassium and cesium units were so small that a second layout for cesium seemed superfluous. Note that the tubes have been bent at the small diameter end to provide for differential expansion between the tubes and the casing and thus relieve the tube-to-header joints of possibly severe thermal stresses.

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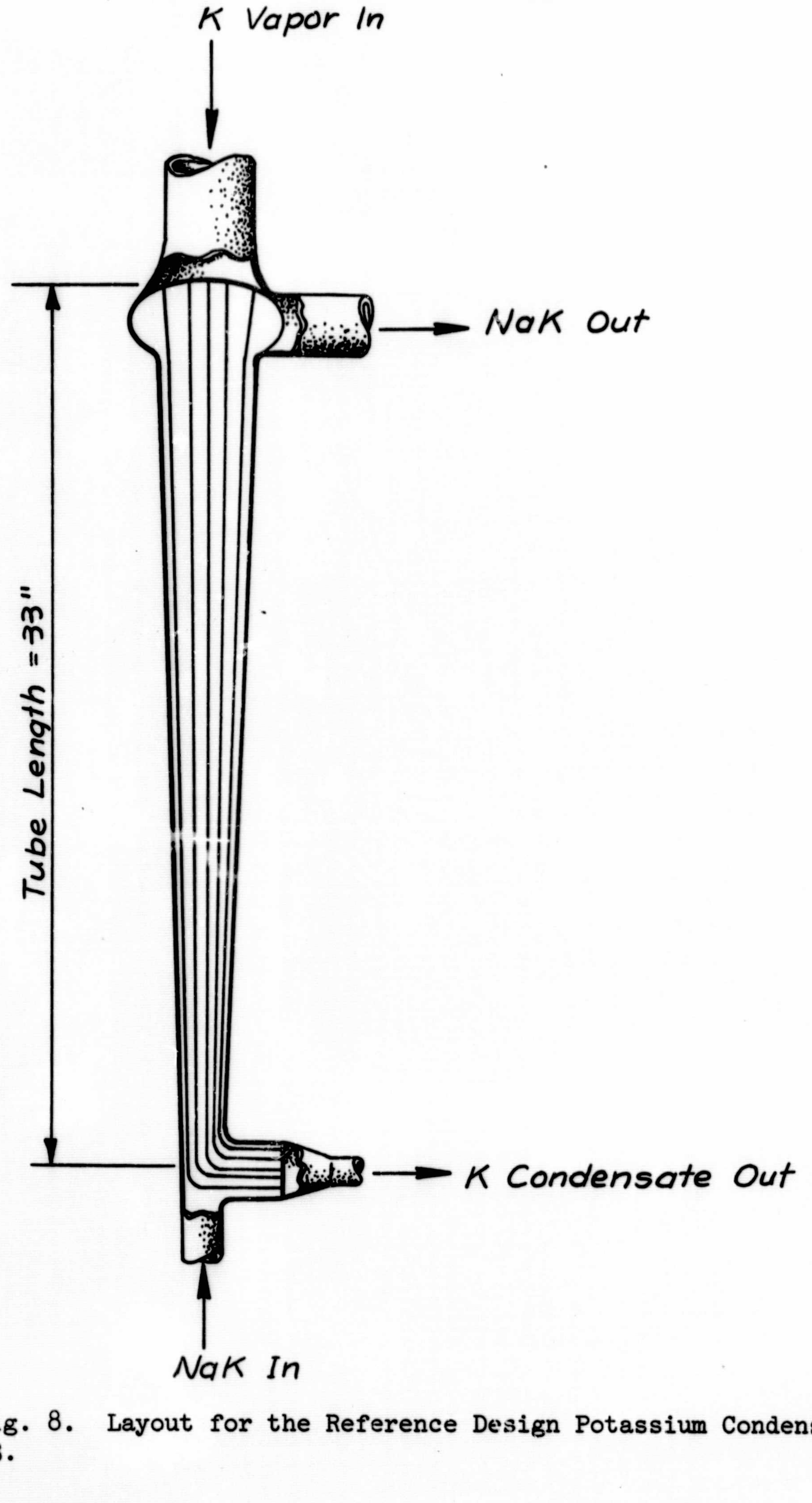

Fig. 8. Layout for the Reference Design Potassium Condenser of Table 3.

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